OPTIMIZATION OF NUSSELT NUMBER IN A HELICAL COIL HEAT EXCHANGER WITH CONSTANT HEAT FLUX BOUNDARY CONDITION USING FLUENT

A thesis submitted by

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This is to certify that the work in this thesis entitled, "**OPTIMIZATION OF NUSSELT NUMBER IN A HELICAL COIL HEAT EXCHANGER WITH CONSTANT HEAT FLUX BOUNDARY CONDITION USING FLUENT**" submitted by **Lalit Ranjan Naik** in partial fulfilment of the requirements for the degree of Bachelor of Technology in Mechanical Engineering, during session 2013-2014 is an authentic work carried out by him under my guidance & supervision.

To the best of my knowledge, the matter embodied in the project has not been submitted to any other University / Institute for the award of any Degree or Diploma.

Date:

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CONTENTS

ABSTRACT

A helical coil heat exchanger has a wide range of application in industries over the straight and shell type heat exchangers because of its greater heat transfer area, mass transfer coefficient and higher heat transfer capability, etc. The relevance of helical coil heat exchanger has been identified in industrial application like turbine power plants, automobile, aerospace, etc. because of above mentioned factors.

The thesis shows the deviation of Nusselt Number and friction factor for different Dean Number ($\frac{D}{d}$ ratio) and Reynolds Number. CFD analysis has been done for varying inlet condition keeping the heat flux of outer wall constant. Copper was used as the base metal for both inner and outer pipe and simulation has been done using ANSYS 13.0. The software ANSYS 13.0 was used to plot the temperature contour, velocity contour and total heat dissipation rate taking cold fluid at constant velocity in the outer tube and hot fluid with varying velocity in the inner one. Water was taken as the working fluid for both inner and outer tube.

NOMENCLATURES

1. A = area of heat transfer (m^2)
2. h = heat transfer coefficient ($Wm^{-2}\ K^{-1}$)
3. p = tube pitch (m)
4. K = thermal conductivity ($Wm^{-1}\ K^{-1}$)
5. Q = heat transferred (W)
6. h= overall heat transfer coefficient ($Wm^{-2}\ K^{-1}$)
7. R_e= Reynolds number
8. N_u= Nusselt number
9. Pr = Prandtl number
10. R = resistance the flow of thermal energy ($W^{-1}m^2\ K$)
11. R_c = pitch circle radius of the pipe (m)
12. L= length of the pipe (m)
13. v = velocity ($m\ s^{-1}$)
14. r= inner radius of the tube (m)
15. V= volume (m^3)
16. α = helix angle (rad)
17. δ = curvature ratio
18. Δ = (temperature) difference (K)
19. μ = viscosity ($kgm^{-1}\ s^{-1}$)
20. ρ = density (kgm^{-3})

CHAPTER-1

INTRODUCTION ABOUT THE PROJECT

Helical coil heat exchangers are always preferred over the straight or shell type because of its compact size and higher film coefficient. They are most commonly used in industries like power generation, nuclear industry, process plant, heat recovery system, chemical process industries etc. Temperature for exothermic reaction of a reactor can be controlled by using these heat exchangers. The design of helical coil exchanger is less expensive. Helical geometry permits the powerful handling at higher temperatures and higher temperature differentials without any exceptionally induced stress or expansion of joints. Helical coil heat exchanger comprises of series of stacked helical coiled tubes and the tube closures are connected by manifolds, which additionally act as fluid in and out area.

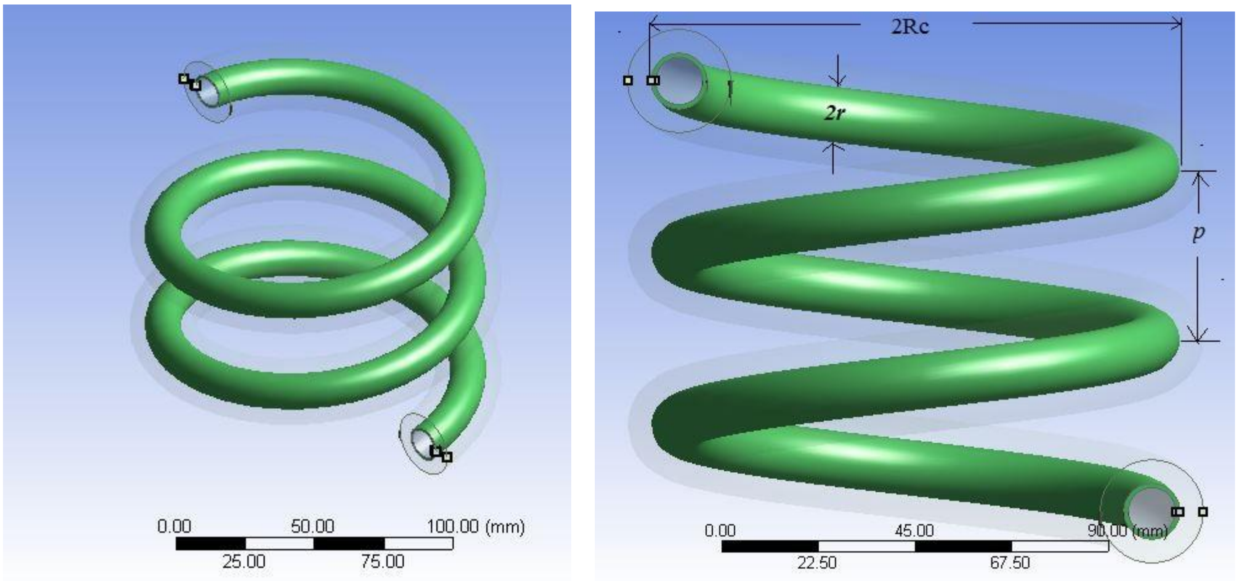


Fig: Schematic diagram of a double helical tube heat exchanger.

The objective behind constructing a heat exchanger is to get an effective method of heat exchange starting with one fluid then onto the next, by direct or indirect contact. Heat transfer occurs in three ways: conduction, convection and radiation. In helical coil heat exchangers the heat transfer by radiation is not taken into account as it is much less and negligible as compared to conduction and convection. Conductive heat transfer can be experienced with a minimum thickness if wall of highly conductive material.in the performance of a heat exchanger, convection plays a major role.

In natural convection, heat transfer occurs by density difference in a fluid due to temperature gradient, and hence doesn’t require any external source like pump, fan or a suction device. Fluid surrounding a heat source receives heat, becomes less dense and rises up. The fluid that is surrounding the hot fluid is cooler and then moves in to replace it.

In forced convection, a heat exchanger exchanges the heat from one moving stream to an alternate stream through the pipe wall. The cooler fluid absorbs heat from the hooter one as the flow is counter flow. If the cold fluid moves along the direction of hot fluid it is called parallel flow and if moves opposite to it, we call that counter flow.

Heat transfer coefficient:

In convective heat transfer heat transfers from one part to another by the movement of fluid particles due to the density difference across a thin film of the surrounding fluid over the hot surface. Through this film heat exchange happens by thermal conduction and as thermal conductivity of most fluids is low, the resistance lies there. The heat transfer by convection is governed by the equation,

$Q=hA\,(t_w-t_{atm})$,

Where, h= film/surface coefficient ($W/m^2 \cdot K$)

A= area of the wall

T_w=wall temperature

T_{atm}=surrounding temperature.

The value of "h" depends upon the different properties of fluid within film region. It is also called as 'heat transfer coefficient'.

The overall heat transfer coefficient can be defined as the overall transfer rate of a combination of series and parallel conductive/convective walls. The 'overall Heat Transfer Coefficient' is expressed in terms of thermal resistances of each fluid stream. The summation of individual resistances is the total thermal resistance and its inverse is the overall heat transfer coefficient, U.

$$\frac{1}{U} = \frac{1}{h_o} + \frac{A_o}{A_i}\frac{1}{h_i} + R_{fo} + \frac{A_o}{A_i}R_{fi} + R_w$$

Where, U=overall heat transfer coefficient

A= wall tube area.

H=heat transfer coefficient (convective)

$R_{f=}$thermal resistance due to fouling

$R_{w=}$thermal resistance due to wall conduction.

Suffixes 'i' and 'o' refer to inner and outer tube respectively

A secondary flow will exist in case in case of helical coil, for which heat transfer rates in this case will be greater as compared to straight tube at the same fluid flow rate.

Aim of present work:

The design of helical coil was done in ANSYS 13.0 workbench. For several D/d ratio the value of Nusselt number was calculated. For different value of input velocity of hot fluid velocity and temperature contour has been plotted. All the work has been done by taking constant heat flux boundary condition with a heat flux value of 30000 Wm^{-2}. Before doing all the calculation, a grid independent test was conducted by taking different values of D/d.

CHAPTER-2

LITERATURE SURVEY

J.S. Jayakumar. According to his study it was attempted to run experimental and theoretical analysis of a helical coiled heat exchanger, in which heat transfer is between fluid-fluid. There exists no previous analysis for helical coil heat exchanger though there are many researches for double pipe heat exchanger. Experimental setup was fabricated to get the output in estimation of heat transfer characteristics, then this experimental data was compared with the CFD calculation using CFD package FLUENT 6.2.

Experimental Setup and Procedure:

The pipe for the construction of helical coil has 10 mm inner diameter and 12.7 mm outer diameter. Pitch of the coil is 300 mm and tube pitch is 30 mm. Material used was stainless steel SS304.

The setup consists of a shell which encloses the helical coil. Cold fluid enters from bottom to top leaving the shell through the nozzle at top. The coil assembly can be replaced if needed.

A tank was provided with electrical heaters to heat the water that to be circulated in helical coil. It consists of three heaters having total power of 5000W. To control the temperature of water at the inlet a controller was connected. A centrifugal pump with ½ HP power rating was connected to pump the hot water in helical coil. RTD (resistance thermometer detectors) were added to measure the inlet and outlet temperatures of the hot fluid and the values are available at the display screen. Cooling water from a constant temperature tank was provided through the shell side and its inlet and outlet temperatures were measured. Its flow was adjusted such that the rise in temperature would not exceed 5^{o}C.

After the temperature attain a constant steady value, by conducting 5 different flow rates through the coil and for three different values of inlet temperature of the helical coil, measurements are taken of the values of flow rates of the hot and cold fluids, temperature at inlet and exit is noted and the power input to the heater and the pump are noted.

These heat transfer characteristics helical coil setup is further studied using CFD code FLUENT. The CFD results matched accordingly with the experimental results within the error limit. A relation was developed to calculate the inner heat transfer coefficient of the helical coil. Based on the results generated under different conditions it may be used to obtain a generalized correlation that may be applicable to other various coil configurations.

A.B. Korane has performed comparative analysis to study friction factor characteristics of shell and helically coiled tube heat exchanger. He continued his studies on two geometries helical coil heat exchanger and square coil pattern having round cross section. Both the coil were constructed by using a 3.33 meter straight copper tube having

10 mm inner diameter and 12 mm outer diameter in 6 turns with pitch 12 mm. the heat exchanger was made by copper tubing and brass connection. Both laminar and turbulent flows were analysed for the Reynolds number range of 886-6200 and having different mass flow rates. The hot water tank with the 3kW capacity thermostatic electric heater was used to pump the hot water through the tubing. The mass flow rate varies from 0.003-0.024kg/s for the hot water which comes from hot water tank. Cold water with the flow rate of 0.003-0.024kg/s is supplied. And the flow rates are controlled by the ball valves provided.

The two helical coils tube side friction factor was determined individually for laminar and turbulent flow. The performance was then discussed according to friction factor and pressure drop.

According to this study he came to the conclusion that

- Performance for the square coil is more than circular helical coil.
- Empirical correlations were developed for both square and circular coils on both laminar and turbulent flow.
- Both the heat exchangers were analysed for laminar and turbulent flow configuration.

The friction factor was minimum for the square coil as compared to circular coil.

Daniel Flórez-Orrego have studied the characteristics of single phase cone shaped helical coil heat exchanger. They conducted experiments on a prototype of cone helical coil heat exchanger with maximum diameter of 15 cm and minimum diameter as 7.5 cm, 3/8 inch pitch and axial length of 40 cm. The flow was in both laminar and turbulent and the range of Reynolds number and Prandtl number were 4300-18600 and 2-6 respectively. According to this study Nusselt number can be found out by $Nu=CRe^{m}Pr^{n}$, where C, m are constants that to be determined and n is the Prandtl number index which is taken as 0.4. An empirical correlation was proposed for average Nusselt number, and it was found that there was a maximum deviation of 23%. Inclination of the velocity vector components in the secondary flow was observed unlike in the straight helical coils. These correlations are not reliable and it failed to give any deviation in Nusselt number due to the tapering and the effect of pitch.

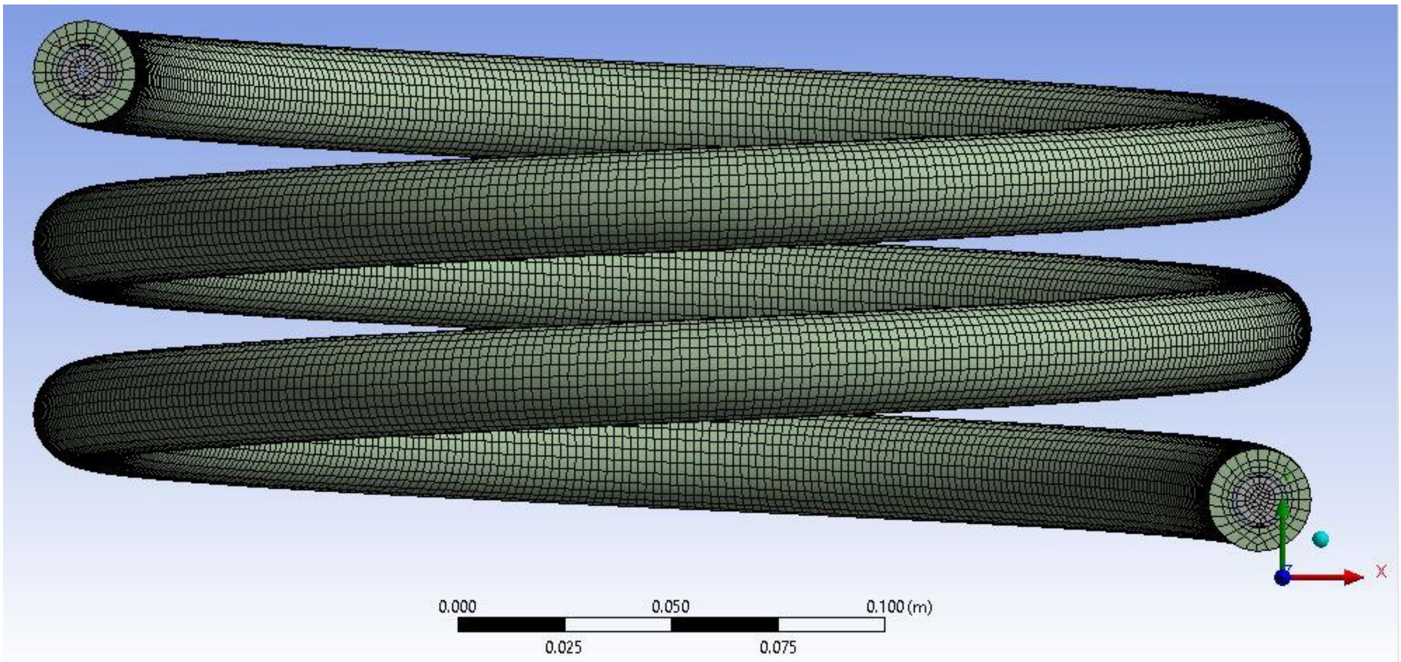
CHAPTER-3

CFD ANALYSIS

3.1. BOUNDARY CONDITIONS

The inlet and outlet conditions have been taken as velocity inlet and pressure outlet respectively. There will be two inlet and two outlet for the flow as this is counter flow. There is a pipe that separates the two flows is made by copper. Copper was taken as the base metal because of its high value of thermal conductivity. The details of all boundary conditions are listed below. Fluid in the inner pipe was taken as hot fluid with 348K and inn the outer fluid cold fluid at 288K. The calculation is done for different D/d ratio with varying inlet velocity of hot fluid and the inlet velocity for the cold fluid was taken constant. The velocity of hot fluid ranges from 1-1.8m/s.

	Boundary condition type	Velocity magnitude	Turbulent kinetic energy	Turbulent dissipation rate	Temperature
Outer inlet	Velocity inlet	2m/s	$1m^2/s^2$	$1m^2/s^3$	288K
Outer outlet	Pressure outlet	-	-	-	-
Inner inlet	Velocity inlet	1-1.8m/s	$1m^2/s^2$	$1m^2/s^3$	
Inner outlet	Pressure outlet	-	-	-	-



3.2 DIMENTIONS

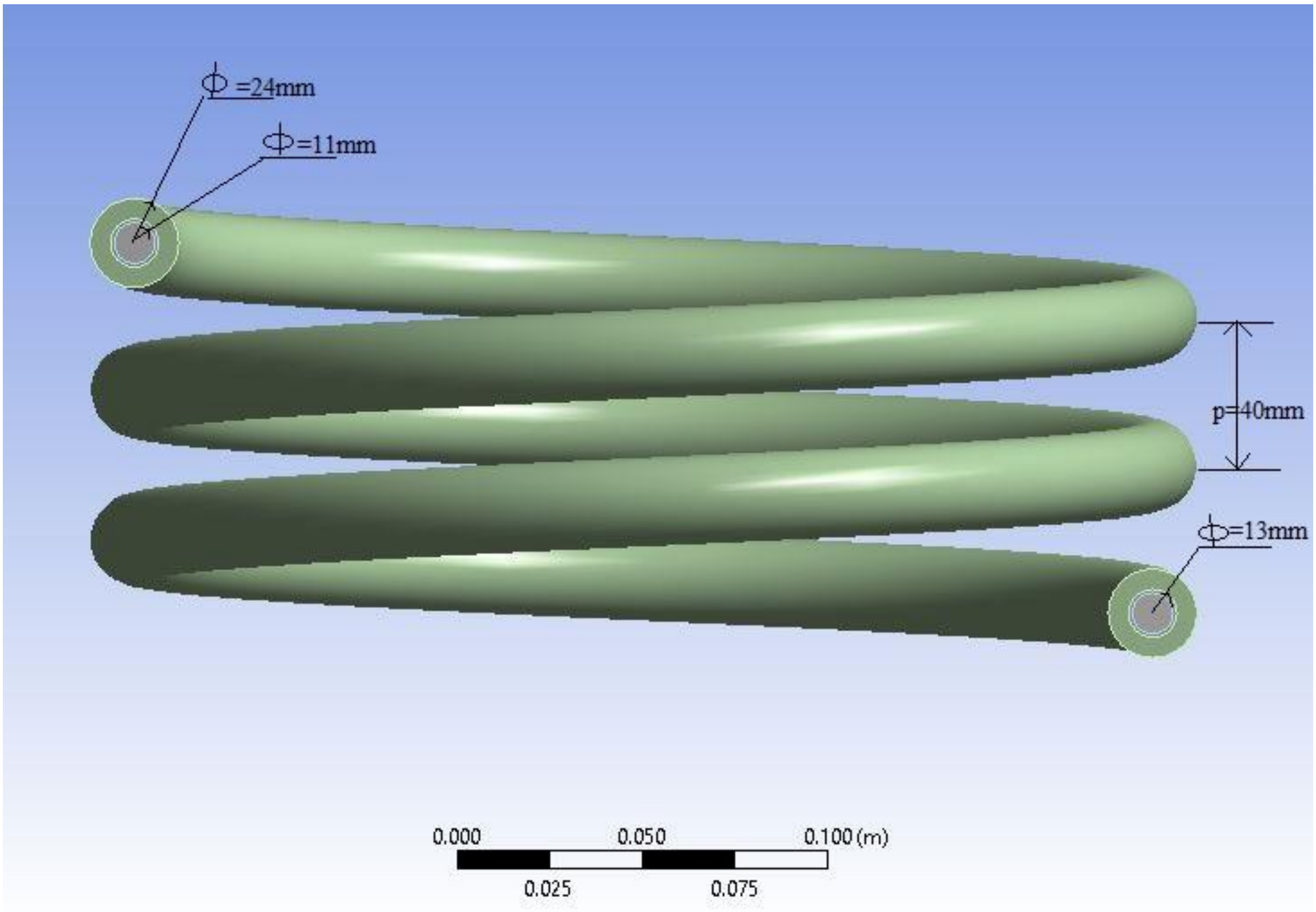
The following figure shows the dimension of the tubes taken.

Diameter of inner inlet= 11mm

Diameter of inner pipe=13mm

Diameter of outer inlet= 24mm

Pitch of the coil= 40mm



CHAPTER-4

RESULTS AND DISCUSSION

4.1. Tabulation

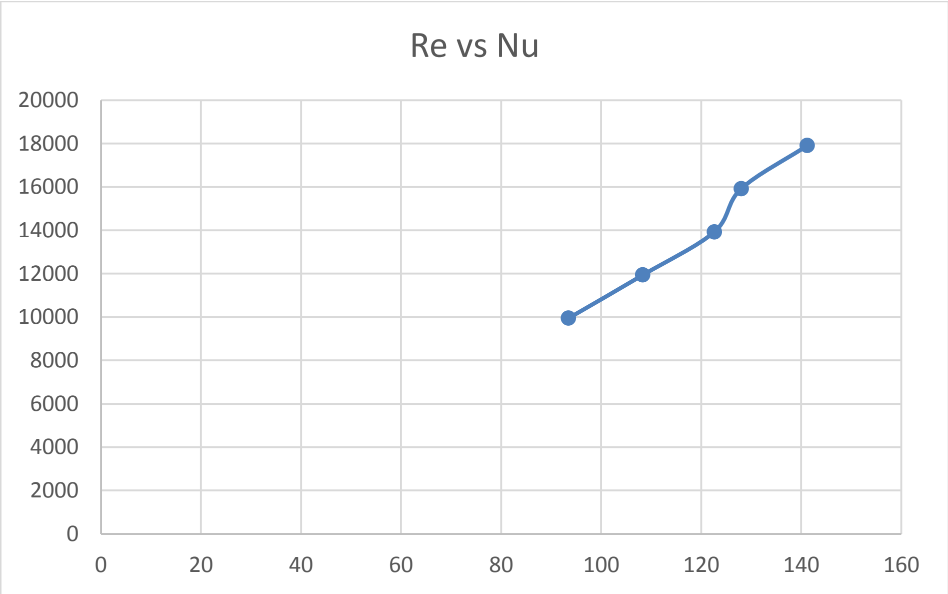
Nu and *f* value for different D/d value

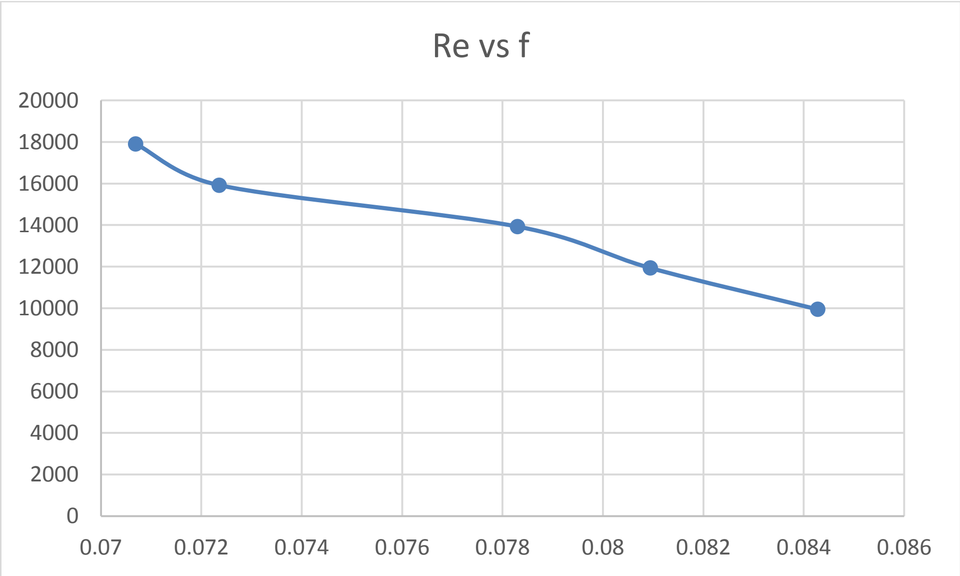
D/d	*V(m/s)*	*Nu*	*f*
10	1.0	93.47928064	0.084277996
10	1.2	108.3163748	0.080944729
10	1.4	122.6052534	0.078300588
10	1.6	127.9894729	0.072354907
10	1.8	141.2002506	0.07068897
15	1.0	91.03997526	0.075594925
15	1.2	105.5822856	0.072638315
15	1.4	119.7608589	0.070301392
15	1.6	127.1549826	0.065779002
15	1.8	140.4391824	0.064430155
20	1.0	84.78380142	0.084078401
20	1.2	98.65265451	0.080609265
20	1.4	112.1652596	0.077850137
20	1.6	125.3826998	0.075587405
20	1.8	138.3429172	0.073684991
25	1.0	84.45707931	0.101717418
25	1.2	98.26866259	0.097373817
25	1.4	111.7242176	0.094021251
25	1.6	124.8848059	0.091168546
25	1.8	137.78868	0.088848845
30	1.0	87.36255815	0.121930938
30	1.2	98.3666604	0.113351466
30	1.4	111.838324	0.109245059
30	1.6	125.0093981	0.10586388
30	1.8	137.9220974	0.103015198

4.2. Plots

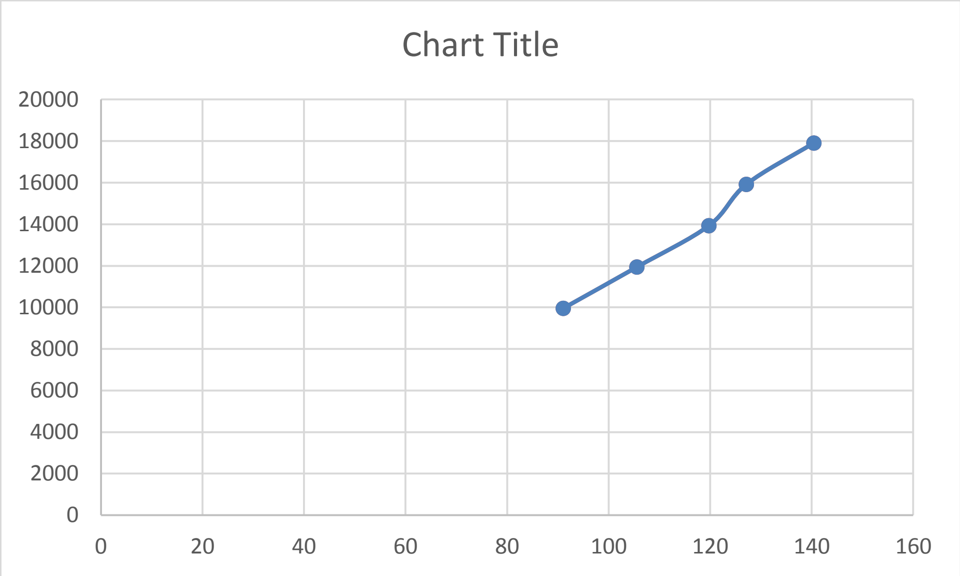
Graphs for Re vs Nu and Re vs f for all the D/d ratios have been drawn in excel and are listed below.

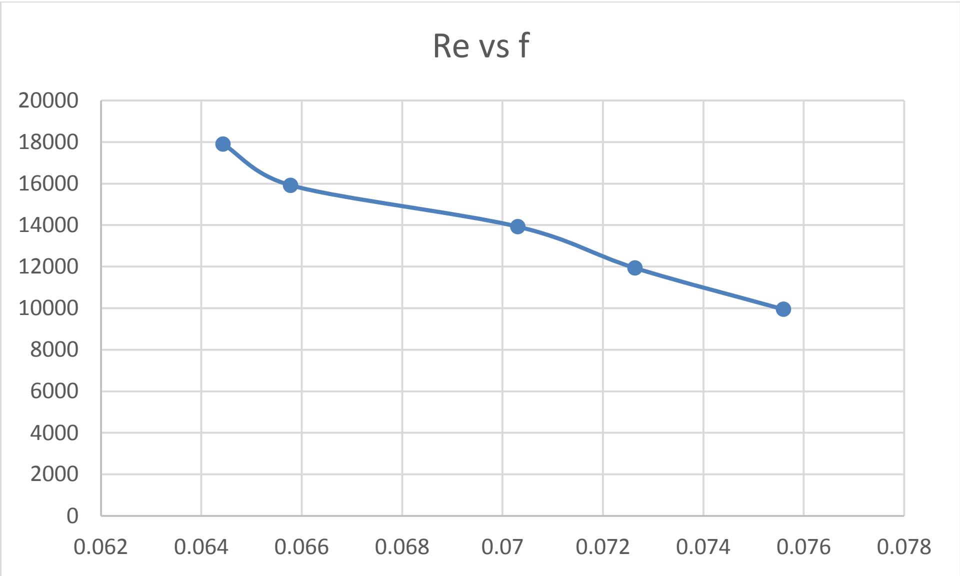
For D/d=10



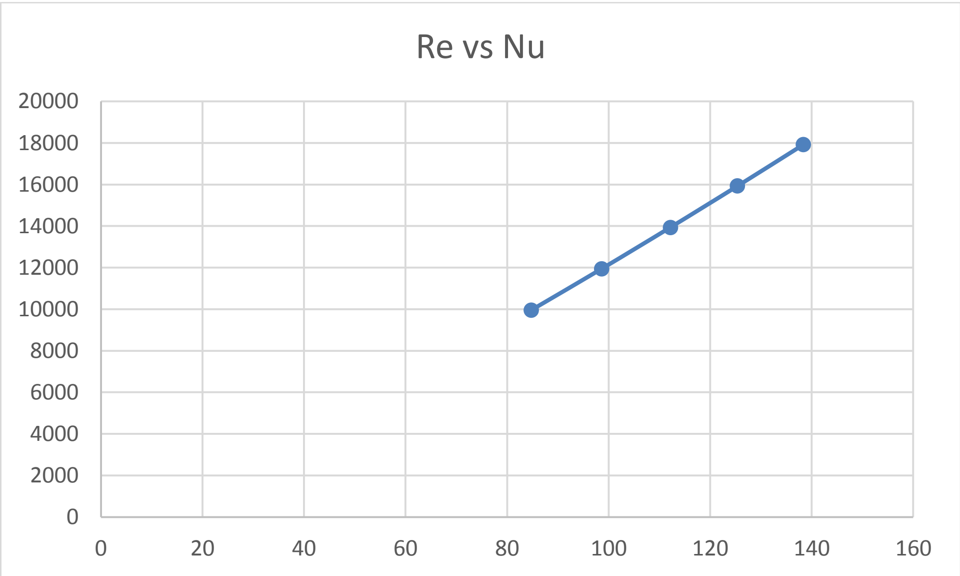


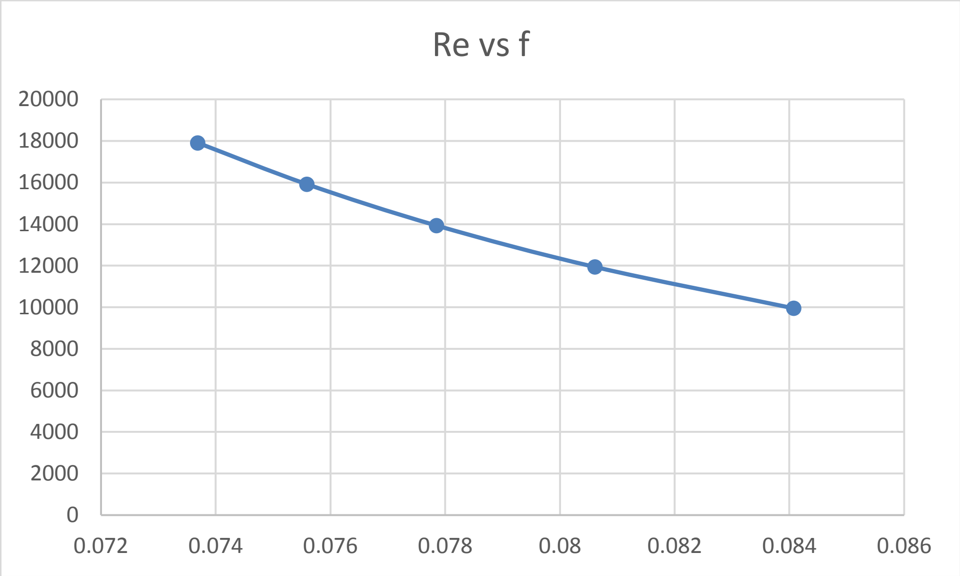
For D/d=15



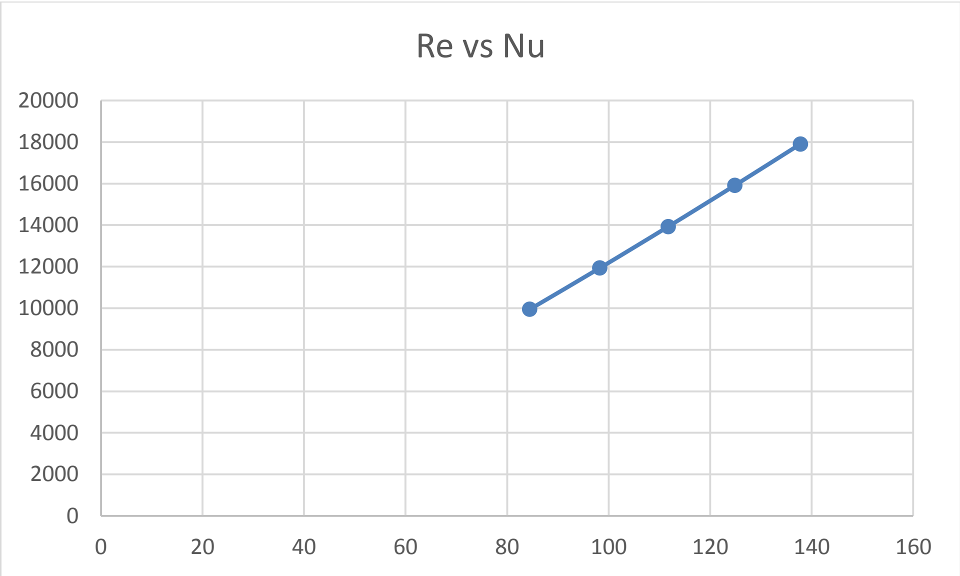


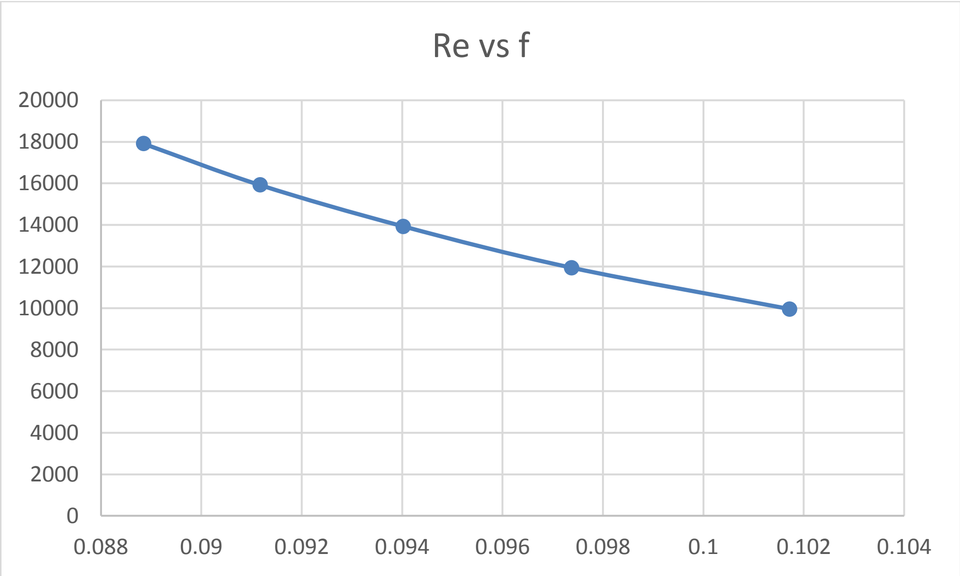
For D/d=20



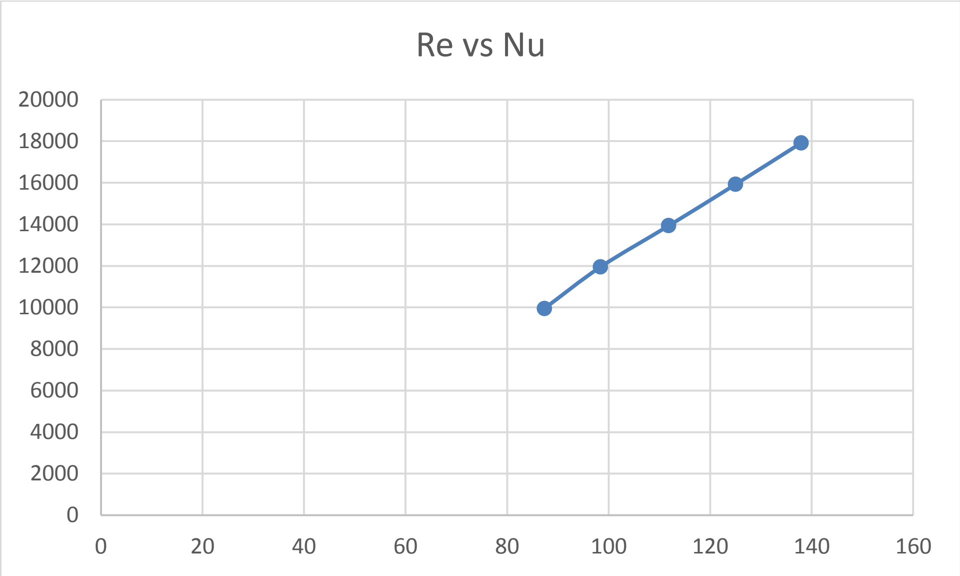

Re vs f
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0.072
0.074
0.076
0.078
0.08
0.082
0.084
0.086

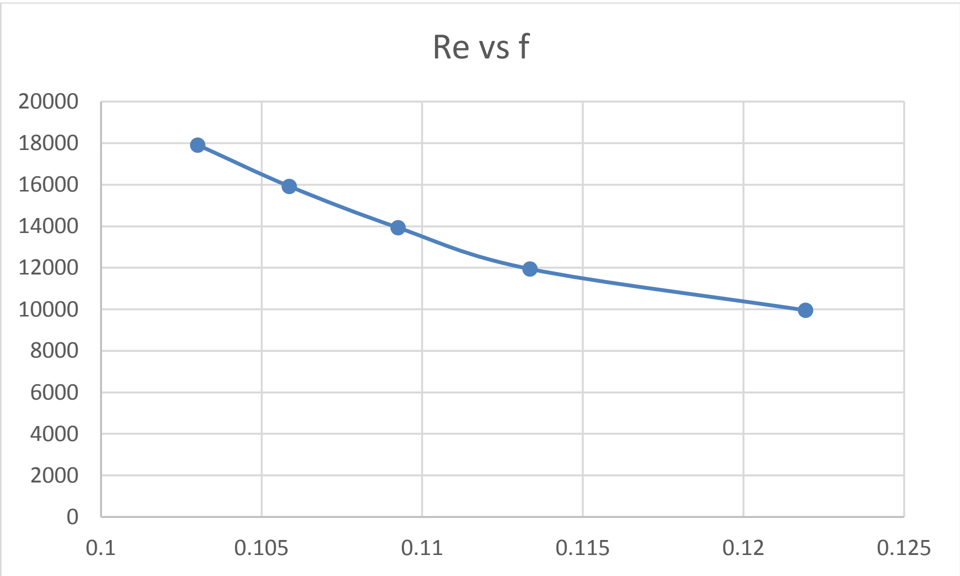
For D/d=25


Re vs Nu
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0
20
40
60
80
100
120
140
160


Re vs f
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0.088
0.09
0.092
0.094
0.096
0.098
0.1
0.102
0.104

For D/d=30


Re vs Nu
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0
20
40
60
80
100
120
140
160


Re vs f
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
0.1
0.105
0.11
0.115
0.12
0.125

CONCLUSION:

ANSYS 13.0 is used for the numerical study of characteristics of heat transfer in a helical coiled double pipe heat exchanger for parallel flow and these results were compared with the experimental results from different study papers and were found well within proper error limit. The study relates the heat transfer performance of the parallel flow configuration and the counter flow configuration. Nusselt number was determined for different points along the pipe length.

We concluded different heat transfer properties at different points along the pipe length in this study like temperature, static pressure, total pressure, kinetic energy etc., for the constant temperature and constant wall heat flux conditions. The velocity vector plot concludes that the fluid particles are undergoing an oscillatory motion inside both the pipes. And pressure and temperature contours shows that velocity and pressure values were higher for outer sides than inner sides of the pipes.

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